

# TURBINE OVERHAUL FREQUENCY— TIME OR PERFORMANCE BASED?

by

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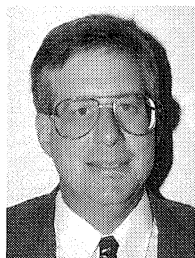
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## ABSTRACT

How long should a turbine run before it is disassembled for inspection and overhaul? This question is increasingly important in today's environment of reduced maintenance and turnaround costs. Many aspects of the degradation in a steam turbine cannot be detected through performance evaluations. Moreover, periodic inspections with a borescope cannot tell the condition of parts of the turbine that cannot be seen.

Two large (60,000 hp and 40,000 hp) ethylene plant turbine drives were recently disassembled for inspection after running over 13 years. During the 13 year run, borescope and visual inspections

were done at each plant turnaround (about every five years). No evidence of erosion was seen on these borescope examinations. In addition, all operational evidence indicated that there was no reduction in the performance of the turbines. Online vibration monitoring showed no problems. The inspection however, showed several areas of severe erosion in both turbines.

The results of these inspections, justification of funds to do the disassembly, and insights into how both jobs were done during a 21 day turnaround window will be discussed.

## INTRODUCTION

The question of when to overhaul a large piece of machinery is one that every machinery engineer faces many times during a typical career. Often, the choice is easy such as when the machine may decide for itself or its performance will degrade so much that there essentially is no choice. But many times the decision is not so clear cut and "conventional wisdom" does little to help:

"Let sleeping dogs lie...."

"Don't rock the boat...."

"If it ain't broke, don't fix it...."

"It's running OK, can't we wait until the next turnaround..."

All of the above ignore the fact that one of the machinery professional's primary responsibilities is to make sure that the equipment runs when it is needed and does not break down unexpectedly. This brings up the question posed by the title of this presentation: Should turbine overhaul frequency be time or performance based? The authors have attempted to insert some data into the literature that will guide others in the future as they are faced with this decision.

The Mechanical Equipment Group at Shell Chemical's OL-5 ethylene plant in Norco, Louisiana, was faced with just this situation when planning started for the 1994 OL-5 turnaround. Since startup in 1981, the plant's two large steam turbines have reliably delivered power with relatively little maintenance. No turbine efficiency loss was ever noticed (although instrumentation needed to measure efficiency is not in place in the field). Normal diagnostic tools available to the Mechanical Equipment Group indicated that all was well within the turbines. Vibration was less than 1.0 mil, reliability was excellent and production has never been limited by lack of horsepower from the turbines. Previous borescope inspections indicate no apparent problems.

The two turbines drive the plant's cracked gas compressor and propylene refrigeration compressor trains and are almost identical to each other. At the time of manufacture, these turbines were the

largest mechanical drive turbines their manufacturer had ever built. Operating conditions are shown in Table 1.

Table 1. Operating Conditions.

	CRACKED GAS TRAIN	REFRIGERATION TRAIN
INLET TEMPERATURE	850° F	850° F
EXHAUST TEMPERATURE	100° F	100° F
INLET PRESSURE	1250 PSIG	1250 PSIG
EXHAUST PRESSURE	4 in HgA	4 in HgA
POWER	60,000 HP	40,000 HP
NUMBER OF STAGES	14	14

## BASIS FOR OPENING TURBINES

The decision to overhaul a large steam turbine is not one that is taken lightly, especially in today's cost conscious plant environment. Ever shrinking maintenance budgets combined with the desire to get a production unit back up and running as soon as possible cause every large job to be questioned.

Since there was little hard evidence of a need to overhaul these turbines, considerable resistance was met to the proposal to inspect both of them. One objection was that opening both turbines at the same time was unnecessary. This argument was countered by the fact that they were both past due for inspection. Also, because of the short duration of the turnaround, by the time the inspection of the first turbine had progressed far enough to show its condition, it would be too late in the turnaround to start the other one, even if major damage was found on the first turbine. The factors discussed later will shed some light on the decision making process.

### History

Although the turbines were running reliably, no industry experience could be found indicating that any users had run large mechanical drive turbines for 14 years without an overhaul and internal inspection. The turbines' manufacturer recommends a major overhaul every five to eight years. This interval is based on their experience with thousands of turbine installations worldwide. Other industry users typically do not go beyond 10 years between major overhauls.

These turbines have approximately 13 years of service with no overhaul. In addition, if they were not overhauled during the 1994 turnaround, they would have had 18 years of service without overhaul by the 1999 turnaround. This amount of service would more than double the manufacturer's recommendation and would be 160 percent of industry practice.

Since the turbines have never been overhauled, there was no way of knowing what effect unique site conditions (steam quality, climate, etc.) might have on their internals. Data from an overhaul are vital to developing a good future overhaul schedule.

### Other Turbines in the Plant

Three turbines of similar vintage had recently been overhauled. The turbines are much smaller, but the 175 psig to 4.0 in HgA steam conditions are similar to conditions on the back end of the large turbines.

One of the turbines had been in service for the same amount of time as the OL-5 turbines and was opened for the first time in February of 1994. Erosive damage was found throughout the turbine. The first three stages had only minor erosive damage and were easily repaired. The last four stages had such extensive erosive damage that a complete permanent repair was not possible within the allotted turnaround time frame. New diaphragms were ordered to install at the next outage. The old diaphragms will then undergo extensive shop repairs.

During the 1990 turnaround, the ethylene refrigeration compressor driver was opened for overhaul and internal inspection.

Previous borescope inspection revealed damage to the second row of blades. Internal inspection also revealed a damaged inlet nozzle. Numerous nozzle trailing edges had broken and passed through the turbine. Although this is a backpressure machine (1250 psig to 175 psig), the conditions of its inlet are the same as the other two turbines' inlets.

In early 1994, Shell's Deer Park Plant shut down its turbine generator set (which is similar in size to the Norco cracked gas train turbine and propylene refrigeration train turbine) for an overhaul after more than 14 years service. The row of blades directly downstream of the extraction nozzle ring suffered particle damage from pieces of the nozzle ring breaking off and hitting the blades. This damage also required a complete reblade of this stage.

Short duration jobs of this magnitude require vast amounts of preplanning. Multiple resources must work out every minute detail on paper several times before the work can be started with confidence. Details that must be worked out before the turnaround starts include:

- Manpower, both hands on and technical support
- Schedules
- Repair shop qualification and selection
- "What if" contingencies (line boring, welding, etc.)
- Tooling and rigging
- Transportation of components, including fabrication of shipping skids
- Spare parts

## DISASSEMBLY AND REPAIR

### Turbine Workscope

Since this was the first overhaul of these turbines, every component was inspected.

- Governor valves rack and servos: Disassemble, clean, inspect, repair
- Rotor: Remove, blast clean, NDT, check runouts and dimensions, check balance
- Diaphragms: Remove, blast clean, NDT, repair as necessary
- Nozzle Box: NDT, repair as necessary, free up seal rings
- Casing: Mechanical clean, inspect all fits, repair as necessary, check casing flatness, check contact between upper and lower halves
- Casing: Remove, clean, lubricate casing expansion keys and pins
- Install new trip and throttle valve
- Remove inlet pipe strain from turbine
- Install wind back oil seals and wiper rings to correct oil leakage on exhaust end
- Overhaul lube oil pumps and drivers
- Overhaul atmospheric relief valve
- Clean and repair leaking tubes in surface condenser
- Renew internal seals as necessary
- Inner casing: Clean, NDT

### Disassembly

Surprisingly, disassembly was relatively easy. Since the turbines had been in continuous service for 14 years, it was thought that many of the fasteners would have to be burned off. This was not the case. All bolts were loosened with the assistance of a 2.5 in drive hydraulic wrench. A few required the addition of heat, but all came loose without galling or other damage.

Contributing to the relative ease of removal are all fasteners are less than 2.0 in diameter and the nuts are coated with a copper based plating. The machine has only four heated studs, these are in the inner case.

Initial observation revealed no real surprises. Stages 11 through the back end of the turbines showed erosion across the diaphragm splitlines and erosion across some sections of the outer casing splitline (Figures 1 and 2). Some areas were near 0.250 in deep and 0.5 in wide. All seals were intact with no signs of severe rubbing. Seal clearances were close to design, but all labys were brittle. Therefore, all seals were replaced.

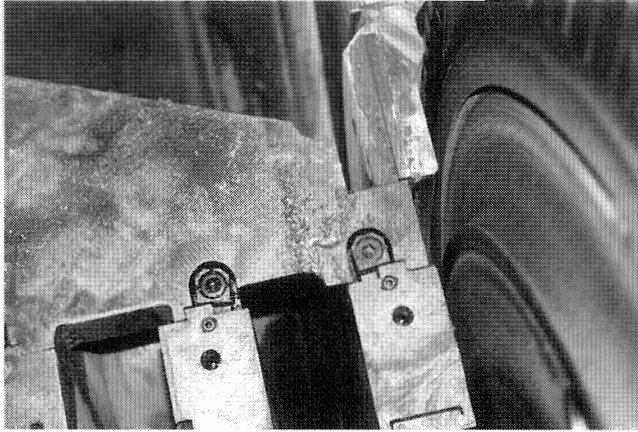


Figure 1. Casing Splitline Damage.

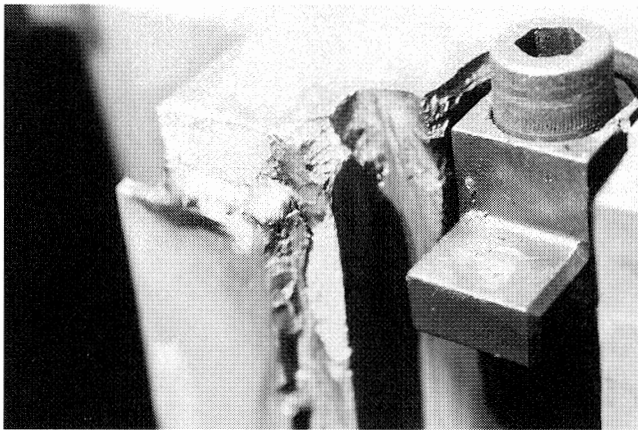


Figure 2. Casing Splitline Damage.

Rotors were in great shape, blading appeared to be in good condition, most packing areas were good except between stage 12 and 13. This area was washed out, pitted, and 0.008 in undersize.

Upon closer visual inspection and the arrival of daylight, damage to the trailing edge of the first stage blades was observed. Each blade had a notch, almost like a saw cut, 1/8 in below the shroud, 3/8 in long, and 1/32 in wide (Figure 3). One turbine, the 60,000 hp unit, showed erosion on the leading edge of the last row of blades, 14B (Figure 4). The erosion was confined to the upper one-third of the blade with 1/16 in of metal removal from the edge. This was not a concern until one blade was found in the same row missing a tenion (Figures 5 and 6). The remainder of the rotor was relatively clean, with no build up of deposits.

Inspection of the nozzle box inlet and valve ports revealed a little lagniappe. A 12 in file, broken into four pieces, was found in various sections of the nozzle box (Figure 7). Fortunately, the file caused no damage to any part of the machine. Apparently, a



Figure 3. First Stage Buckets-Trailing Edge.

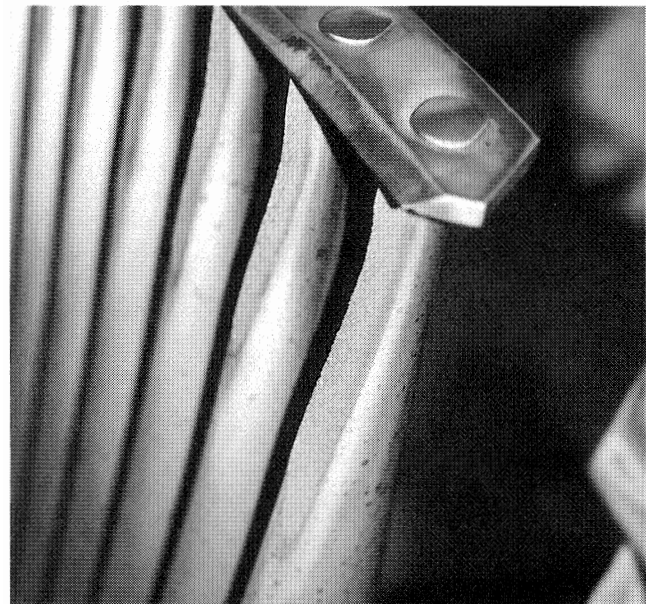


Figure 4. Stage 14B Erosion Damage.

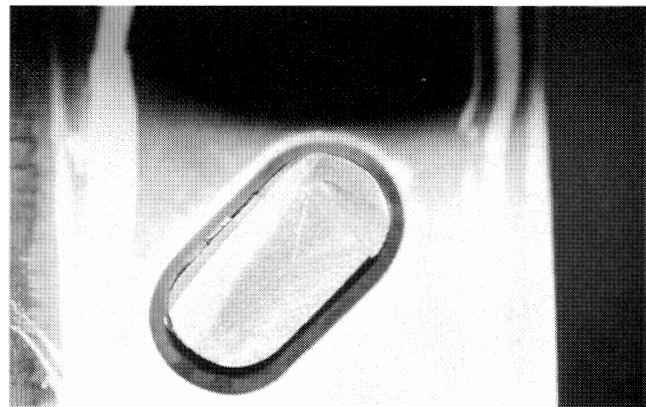


Figure 5. Failed Tenion on Stage 14B.

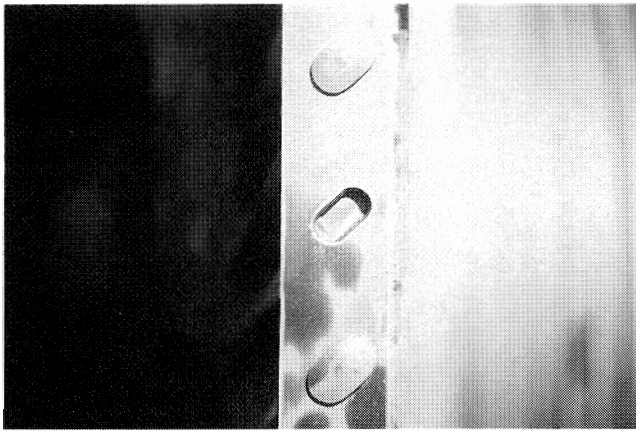


Figure 6. Failed Tenion on Stage 14B.

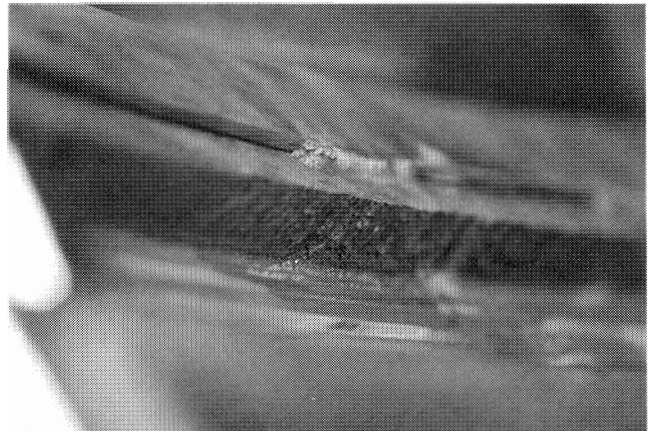


Figure 8. Diaphragm Pocket Sealing Surface Damage.

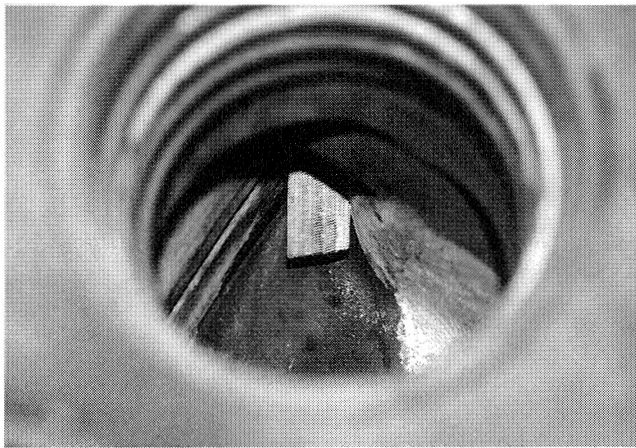


Figure 7. File Found in Inlet Nozzle Port.

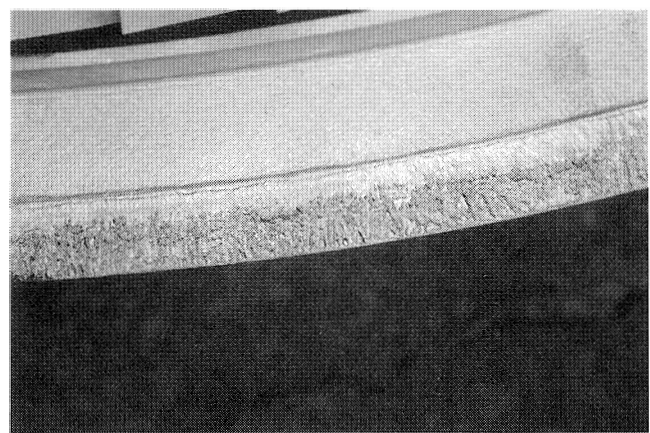


Figure 9. Diaphragm Sealing Surface Damage.

craftsmen had used a file as a gasket scraper during a previous maintenance of either the valve rack or trip valve. It appears the file was broken by a governor valve closing down on it.

After rotor and diaphragm removal, significant damage was found. The casing grooves from 11 through 14B had heavy wash out on the sealing surface. Erosion was typically 1/8 to 1/16 in deep and evident 360 degrees around the groove (Figure 8). The diaphragm sealing surface had a matching pattern (Figures 9 and 10). The 360 degree nozzle box had numerous partitions where the trailing edges had broken off. Evidence of these pieces impacting the blades could be found on the first four rows of blades. The bottom center of the nozzle ring had two partitions completely missing (Figure 11).

#### *Summary of Internal Damage Found*

- Nozzle ring missing pieces of trailing edge
- Nozzles completely missing
- First row rotating blades cut on trailing edge
- Last row of one rotor, slight erosion, missing one tenion
- Diaphragm splitline erosion stages 11 through 14B
- Casing splitline erosion from stage 11 on, erosion between low pressure packing sections
- Diaphragm case groove and diaphragm seal fit heavy erosion and wash out, stages 11 through 14B
- Rotor interstage seal washout between stage 12 and 13



Figure 10. Diaphragm Sealing Surface Damage.

#### *Casing and Diaphragm Damage*

Inspection of the high pressure casing revealed no damage. Steam erosion was, however, present in the low pressure casing at the 11 through 14B stages. The erosion damage was confined to the casing splitline near the diaphragm pockets and to the sealing faces of the diaphragm grooves on those stages. As noted earlier, damage in some spots was quite deep, especially near the inside of the splitline.

It is likely that thermal stresses caused the interior of the casing to open slightly. During thermal transients, such as startup and



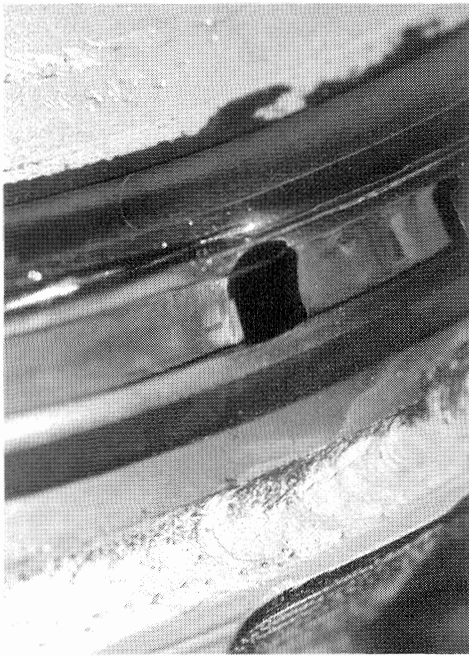


Figure 11. Inlet Nozzle Ring Damage.

shutdown, the casing ID heats at a faster rate than the OD. This thermal gradient tries to make the ID expand however it is constrained by the OD. If the gradient is large enough, the ID is forced to yield slightly. Once the thermal gradient is reduced, and thermal stresses are normalized, the splitline inside the bolt pattern can open slightly. Normal casing material is no match for this leakage path combined with wet steam.

Diaphragm damage was as expected: the diaphragm splitlines and sealing surfaces of the diaphragm pockets (Figures 12, 13, 14, and 15). Most of the diaphragm alignment keys had to be repaired or replaced.

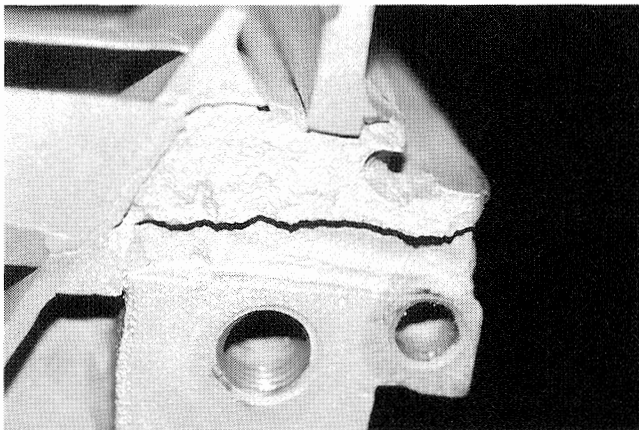


Figure 12. Diaphragm Splitline Erosion.

#### Casing Repair

The damage that was found made it obvious that repairs had to be done. So now the question was: How should the repairs be made? The main requirement here was that all repairs had to be made without affecting the schedule. This meant that removal of the casing to a shop was out of the question; repairs would have to be made in the field. For the same reasons, the field repairs had to

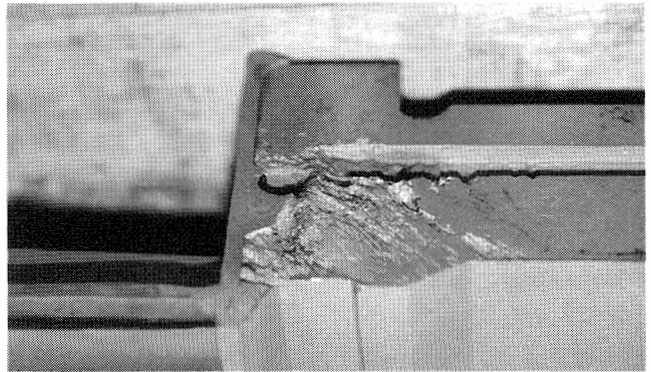


Figure 13. Diaphragm Splitline Erosion.

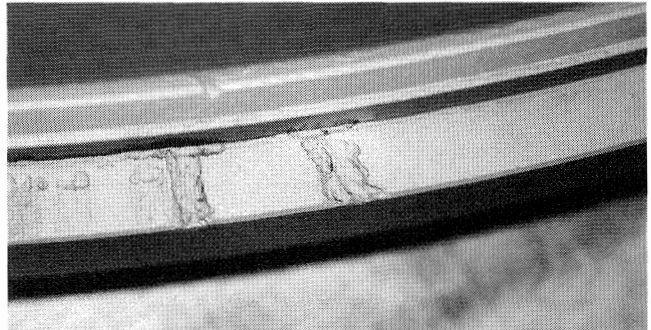


Figure 14. Diaphragm Sealing Surface Damage.

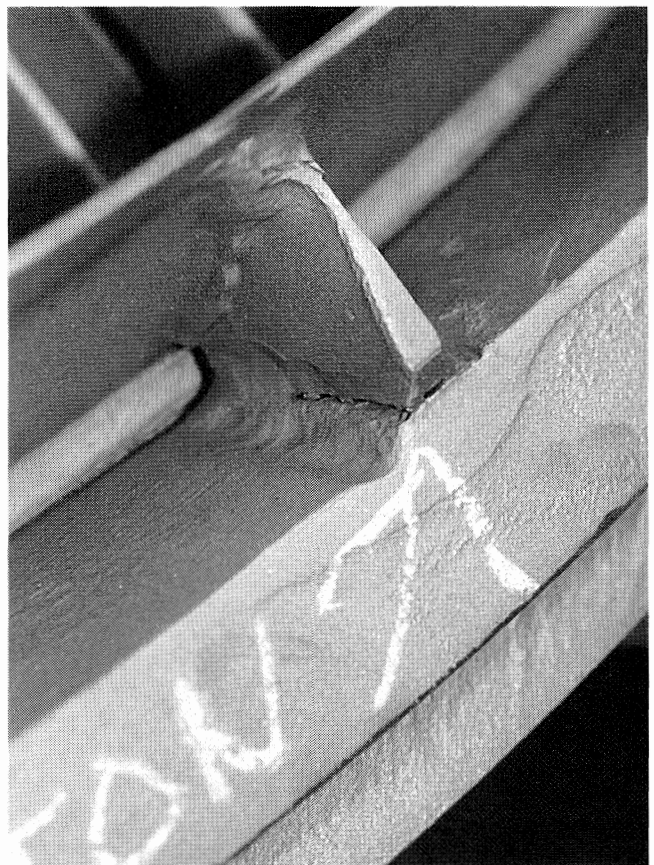


Figure 15. Weld Erosion on Twelfth Stage Diaphragm.

be accomplished without excessive heating of the casing. This could have caused warping of the splitline, necessitating time consuming field machining.

The repair technique employed was as follows:

- Grind out the eroded areas to suitably prepare them for welding (Figures 16 and 17). Peen the perimeter of weld to raise metal, preventing a visible fusion line.
- Weld repair these areas using a tungsten inert gas (TIG) process and Inconel filler material. Some areas in the low pressure section were repaired using 7018 rods (Figure 18).
- Hand work the repair areas to reestablish a flat splitline. A finished area is shown in Figure 19.

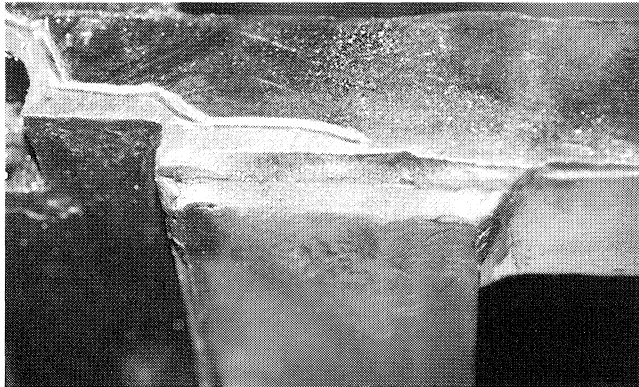


Figure 16. Casing Splitline Weld Prep.

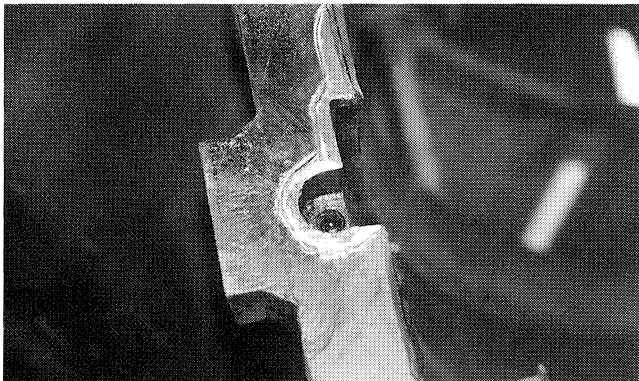


Figure 17. Casing Splitline Weld Prep.

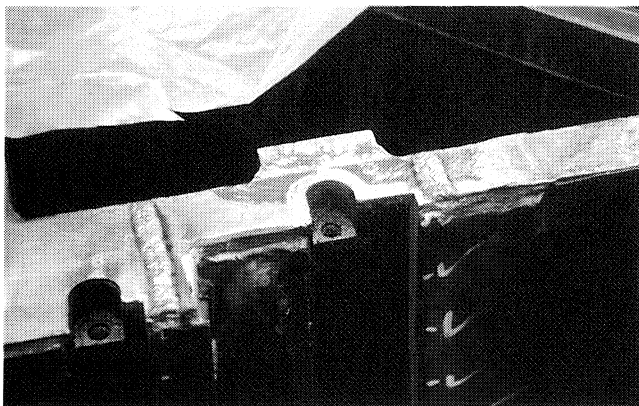


Figure 18. Casing Splitline Welding.



Figure 19. Typical Finished Casing Splitline Repair.

This repair technique worked very well. Splitline flatness was verified by a "blue check" of the casing halves after repairs were complete. This check showed that the objective of a flat splitline was indeed achieved.

A word of warning: because of the amount of handwork involved in this type of repair, highly skilled craftsmen are needed to do the work. The previous experience of the contractor and individual craftsmen should be thoroughly evaluated before beginning work.

#### *Problems With Boring the Case*

As soon as the diaphragms were removed, it was obvious that some of the casing grooves would require field machining. Minor erosion was expected but nothing of the magnitude observed. Arrangements were made with a contractor who supposedly had two bars available that could span the 22 ft between bearing housings. Having two bars would allow the boring work to proceed on both turbines at the same time.

Typical repair techniques were evaluated based on technical merit and amount of time required. Had more time been available, the repair method may have been to undercut the grooves, weld overlay with a more erosion resistant material, and recut to original dimensions. This procedure has the potential for warping the case along with consuming considerable time.

Undercutting the grooves and installing bands that could be mechanically attached to the case was also considered. Due to the amount of material that would be removed from the case and time involved, this proposal was also rejected.

Machining of the case grooves until 100 percent cleanup was the repair process selected. Patch rings installed on the diaphragms would make up the difference in groove width so that the diaphragms would remain in the same axial position. This allowed only one setup of the boring bar, removed the least amount of metal from the casing, and did not subject the case to extensive welding.

The only problem was in the 14B casing groove. Due to the configuration of the casing, there was not enough space to get the cutting head into the groove. The erosion damage on 14B was not as bad as the others and confined to a 90 degree arc, 45 degrees on each side of bottom center. Since this diaphragm sees a low differential pressure and low temperature (less than 200°F) a cold repair technique was chosen. This consisted of mechanically cleaning the groove and filling the erosion areas with an epoxy type compound. The repaired areas were hand dressed to match existing case grooves. The next major overhaul, probably in 10 years, will allow an evaluation of the technical merits of this repair.

Case boring was relatively straightforward. The boring bar was installed in the case with center reference provided by the bearing housing bore. Spider bushings supported the bar near the cutting area. After preliminary setup, the upper casing was installed and bolted down. Machinist access to the inside was provided through the valve rack flange and manways in the exhaust casing. The boring bar setup is shown in Figure 20.

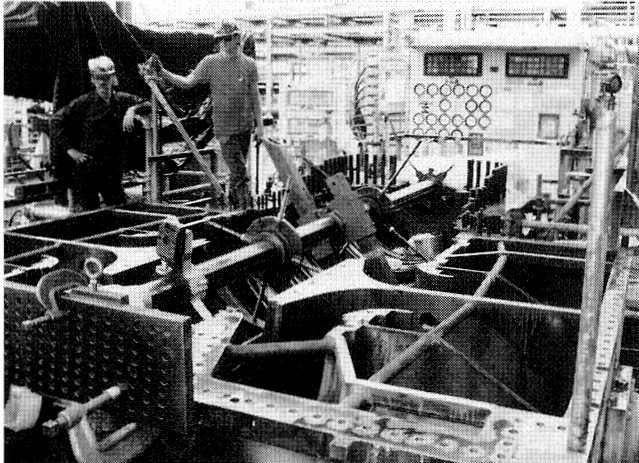


Figure 20. Boring Bar Setup.

After completion of the boring job, one turbine looked great with everything square and parallel. However, the other turbine had casing grooves that were not square to the adjacent groove and were of varying widths. Investigation revealed the boring contractor had one bar that would span the bearings and another that was shorter. This short bar is the one that produced the inaccurate machining. The grooves were recut with the long bar and everything came out correct.

It should be noted that the same crew made the same error on the Deer Park turbine eight months later. It continues to be the authors' opinion that an accurate boring job cannot be accomplished unless the boring bar can reference both bearing housings simultaneously.

#### Diaphragm Repair

Most of the diaphragms in the low pressure end of the turbine needed some type of repair work. As with the casing, the stage 11 to 14B diaphragms required the most repairs. Some of the diaphragms were damaged enough that they would have been replaced had replacements been readily available. Since replacements were several weeks away, a suitable repair method had to be developed. A cooperative effort among all parties involved (turbine manufacturer, Shell, repair shops) was used to develop a repair method for each diaphragm.

The diaphragms needed repairs in two principle areas: the splitlines and the sealing surfaces. Some of the diaphragms also needed repairs in the interstage seal hook fit areas.

#### Diaphragm Splitline Repairs

The splitlines of the diaphragms were heavily eroded and had to be repaired to restore the sealing areas and the alignment keys. Repairs were made by:

- Machine the outline (sealing area) of the splitline surface to a depth of approximately 0.125 in to prepare for welding (Figure 21). Also prepare alignment key groove for welding if necessary.
- Weld repair with Inconel material.

- Machine splitline surface to establish a flat sealing surface (Figure 22). Many diaphragm leveling screw holes had washout in the threads. These were bored out, filled with weld metal, drilled and tapped to original dimensions.

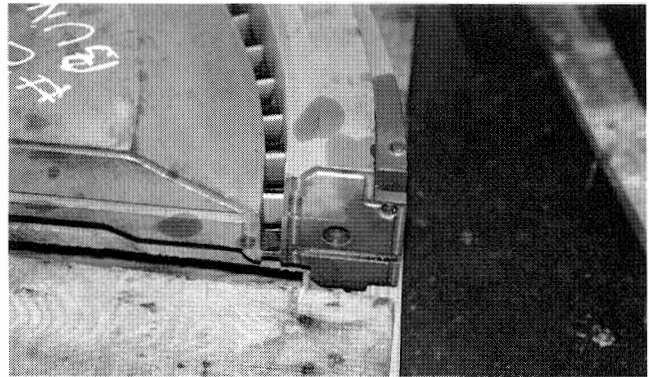


Figure 21. Diaphragm Splitline Repair Weld Prep.

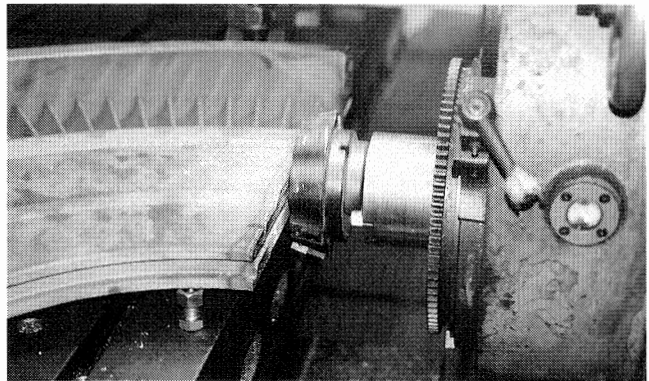


Figure 22. Machining Diaphragm Splitline.

#### Diaphragm Sealing Surface Repairs

The sealing surfaces on the axial face of the OD of the diaphragm had to be repaired on stages 11 to 14B. Complete weld repair was not practical within the time available due to the possibility of warping the diaphragms. A mechanical repair method was therefore chosen (Figure 23).

- The sealing surface of each diaphragm was machined back a suitable depth to allow for cleanup of the surface and adequate thickness of the patch ring.
- A patch ring was made out of carbon steel plate and attached to the diaphragm by bolting. The two shops took different approaches to making the patch rings. One shop rolled the rings from barstock while the other cut the rings from plate. Both methods worked equally well and took about the same amount of time. Lesson here: don't get so stuck on one repair method that you don't consider others.
- The bolts were counter sunk and covered by weld metal. To avoid welding directly on the head of the capscrews (and thus weakening it), a washer was placed on top of the bolt head, then weld buildup on top of the washer.
- After the diaphragm grooves were field machined to final size, the diaphragm patch rings were machined to match the casing groove.
- The axial crush pins were welded up to allow for hand fitting in the field. The pins were machined 0.005 in wider than the casing groove. This made field fitting the crush pins much easier.



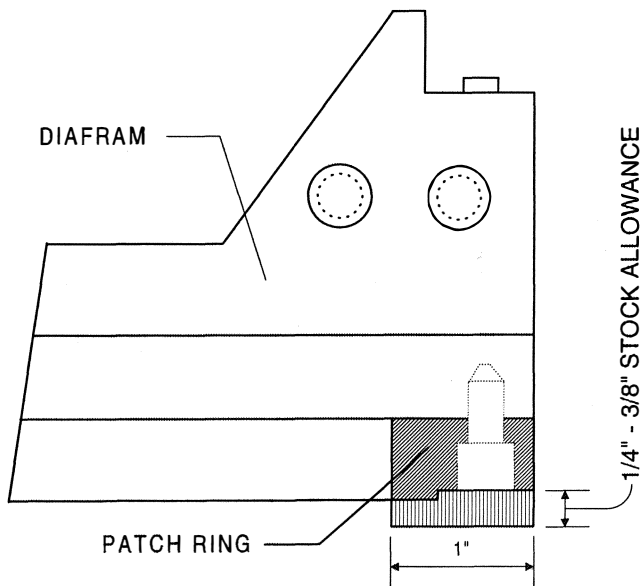


Figure 23. Diaphragm Mechanical Repair Method.

#### Nozzle Box Repair

The nozzle box is built such that the outer casing is not subjected to high pressure inlet steam. The high pressure is contained within the nozzle box, the nozzles convert the high pressure steam to lower pressure, high velocity steam. Thus the casing sees about two-thirds the inlet pressure. Six governor valves control steam flow into the 360 degree nozzle ring. The nozzle box is shaped like a doughnut with the six nozzle passages integrally cast. Nozzle ring blades or partitions are fabricated independent of the nozzle box. The completed nozzle ring assembly is then welded into the nozzle box. While this design is very functional, it is somewhat hard to repair. Nozzles cannot be removed and renewed.

Repairs began with a thorough blast cleaning of the nozzle box casing. Machined areas and the nozzles were masked off to prevent damage. These areas were hand cleaned. NDT examination of the case revealed several cracks (Figure 24) on the side opposite the nozzle ring. These cracks were located in and around welds made to seal up coring plugs. Excavations of these cracks indicated they went all the way into the interior. The cracks were excavated until removed and then carefully weld repaired. Due to numerous machined surfaces on the nozzle box, it was highly desirable to weld repair in such a way that post weld heat treatment was not required. This was accomplished through careful selection of filler material and weld techniques.

The trailing edge of nozzle partitions is very thin. Foreign particles and fatigue combine to cause some of these trailing edges to break off. Approximately one-third of all of the partitions had pieces missing. In addition, the two nozzles at bottom dead center were missing completely with the appearance of being blown out. Damage to the bottom two nozzles has yet to be explained.

Repairs to the trailing edges is somewhat routine for welders experienced with this type of specialist work. The rough edges are hand dressed, weld material is added to reconstruct the blade, and the blade profile and spacing are re-established through detailed hand dressing. Replacement of the two missing vanes was another story. This highly skilled task involved smoke, mirrors, and a good dose of magic. Nozzle box repairs were completed in the OEM's shop in less than a week.

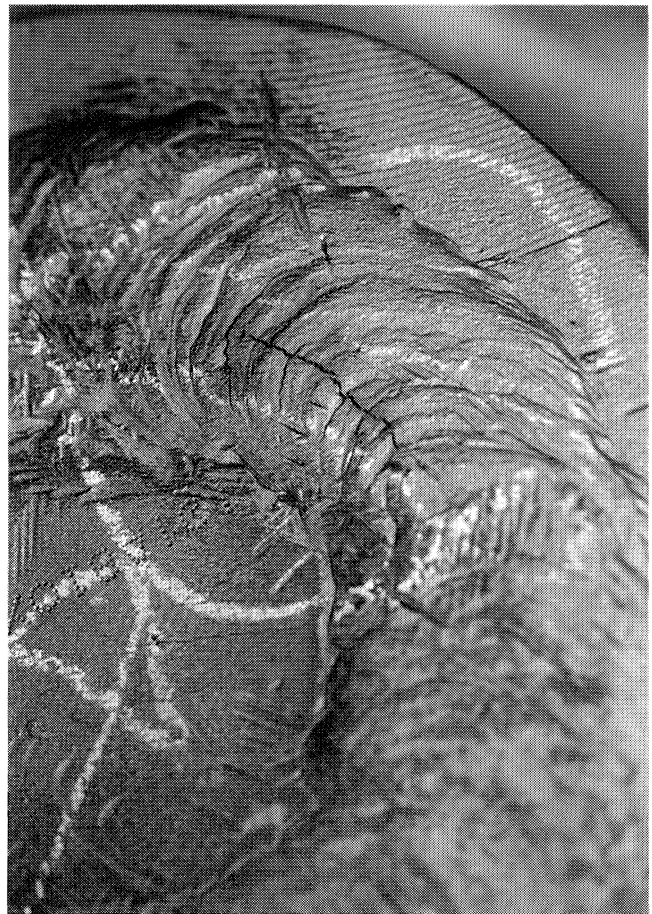


Figure 24. Crack in Nozzle Box.

#### Rotor Repair

After careful evaluation of both rotors, one of the rotors had to be selected for repair and reinstallation. One rotor needed blades on the first and last rows while the other needed the first row of blades only. Since the latter was in better shape, this was the obvious choice. But, there was still a problem; no first stage blades were available from stock and only 10 days were left before the rotor was needed to fit the diaphragms.

Panic stricken calls to the turbine OEM produced a commitment to manufacture and install new first stage blades within 12 days. Needless to say, there was much skepticism of such a commitment and even more surprise when it actually took place. The blades were manufactured at the OEM's blade manufacturing facility and installed by its Dallas service center. The rebladed rotor was low speed balanced and returned to the jobsite. During the reblading process, the rotor that needed both first and last stage blades was used to set the diaphragm heights. Thus, no time was lost in the reassembly process.

#### Spare Rotor Repair

After the turnaround, the spare rotor required the following work:

- Reblade first and last rows
- Roll journals to remove scratches
- Skim cut backside of eleventh stage disk to remove corrosion/erosion pitting (stress analysis required to ensure disk integrity)



- Hone steam balance holes to remove stress risers (from erosion)
- Install redesigned eleventh stage locking piece
- Various minor cosmetic repairs
- At speed balance

Two repair options were offered to repair the 12 in journals: roll to improve surface finish, or undercut and weld to correct size. The rolling option is considerably less expensive. However, the resulting journal size would be approximately 0.003 to 0.005 in undersize. With a 12 in journal and 0.024 in clearance on the four pad tilt pad bearing, an additional 3.0 mil of clearance did not seem to matter. To be on the safe side, a rotor response analysis was conducted. This showed the rotor to be very stable and basically unaffected by the increased bearing clearance. Therefore, this option was selected.

Shaft interstage packing areas on the wet end of the rotor all had pitting and erosion and were slightly undersize. Repair options considered were: undercut and make nonstandard packing or weld overlay to design size. After consultation with the OEM's engineering department, it was determined the extra clearance in this section of the turbine would have a negligible effect on power and efficiency. Therefore, a third option was selected, leave as is. It is anticipated after the next run cycle these areas will require weld repair.

Undersizing the areas and making nonstandard packing, although technically acceptable, was not considered due to the shared nature of this rotor. As stated previously, one rotor is shared between four machines at two different locations. When the rotor becomes a "special," interchangeability begins to suffer. This is compounded by the fact that since overhaul intervals are so long, 10 to 15 years, the mechanical staff will probably be different each overhaul.

#### DEER PARK TURBINE AFTER 16 YEARS IN SERVICE

Eight months after the Norco turbines were overhauled, the sister unit in Deer Park, Texas also came down for a scheduled outage. Based on observations from the Norco unit, it was decided to go into the highest horsepower turbine at Deer Park. New diaphragms for stages 11 through 14 were ordered and delivered prior to the outage. After the case grooves were machined, the new diaphragms were cut to fit. The new diaphragms greatly reduced repair time.

Essentially, this rotor was in the same shape as the Norco rotor, only slightly worse in some areas. The first row blades had the same erosion on the trailing edge, the last row were in good shape with no problems. The biggest difference was in the packing area condition.

The first set of high pressure packing fits inside the nozzle box doughnut and is a alternating high-low style. Apparently, the packing was incorrectly positioned during assembly and the packing high teeth rubbed the sides of the rotor high teeth, reducing the width of the rotor lands by 50 percent. Since this packing area has a high delta P, and can bypass a lot of steam, this area had to be weld repaired to restore design conditions.

The wet end interstage packing areas also had considerably more erosion damage than the other rotors. Some areas were washed away such that 1/8 in of rotor was missing per side. The low pressure packing area between the sealing steam inlet and the last wheel also was undersize and the edges of the packing grooves were rounded over from erosion. It was decided to weld repair all of these areas.

In addition, the high pressure sides of stages 12, 13, and 14A had severe pitting and erosion on the disks. After a detailed stress analysis, the areas were skin cut (up to 0.190 in) to remove all stress risers. The low pressure sides required replacement of the balance weights and enlarging of the balance weight grooves.

Considering the service and operating life, the required rotor repairs were considered to be relatively minor. If the rotor repair costs are divided by the years of service, maintenance is a very low percentage of the maintenance budget.

#### CASING SPLITLINE GAP IN THE LOW PRESSURE SECTION

The inner exhaust casing was open at the splitline. This casing has no bolts holding it together and has apparently opened up from thermal stresses. The inner casing separates the double flow stages from the condenser inlet.

The blue check of the splitline and visual observations revealed a significant gap in the splitline at the stage 14A/14B flow splitter (Figure 25). There was no contact and, in fact, a 1/16 in to 1/8 in gap was visible. Unfortunately, this gap was not discovered until about half of the available maintenance time was used. With not enough time left in the outage to do anything about it, the decision was made to leave it as is. It can be seen from Figure 25 that there was no erosion damage in this area. This area of the turbines will be considered for installation of some internal bolting at the next outage.

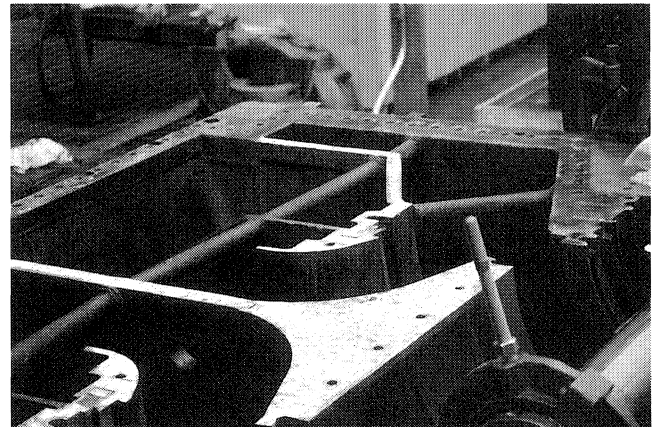


Figure 25. Blue Check of Splitline.

#### Summary Of Lessons Learned

Several things were learned from this outage that could be done differently to reduce the outage time in the future.

- Since the diaphragms on the wet end of the machine required so much repair, new last stage diaphragms were ordered for the Deer Park turbine. The new diaphragms were oversize in the casing groove and cut to size after the case was remachined. This saved a considerable amount of time. If new diaphragms aren't practical, then filler strips can be prefabricated prior to the outage.
- Better overview of casing boring operations are a must.
- Use of the OEM field representatives were beneficial in timely completion of the outage by providing direct access to factory personnel and data.
- Cooperation between the OEM and two local shops, one of which is the local shop of a different OEM, can be achieved by proper planning and communication. This cooperation was beneficial to all parties.

Erosion damage can best be detected by overhaul and internal inspection including complete disassembly. Most of the diaphragm damage discovered in this case would not have been seen had the diaphragms not been removed from the case.

## CONCLUSIONS

The details of the damage and repair are presented above to give the reader a graphic demonstration that a turbine can be:

- Very reliable
- Perform up to the needs of the operations department
- Borescope inspected
- Vibration monitored

but still be in extremely poor condition internally!

The justification for this work consisted of pointing out the history of these turbines and what little data that were available from other users along with the kinds of damage that could occur in the turbines. This damage might not be detectable through normal borescope internal inspection or a change in the performance of the turbines. All of this explanation was very helpful but the argument that carried the day was that the reliability of the turbines would not be good enough to guarantee another five year run. Expected damage included erosion of case fits and diaphragms, diaphragm splitline erosion, air foil, and nozzle ring damage.

### *Borescope Inspections*

These turbines have been inspected by the OEM using a borescope at each unit turnaround (1985 and 1990). None of the borescope inspections indicated that any erosion was present inside of the turbine. This is not meant as a knock against borescope inspections, they are a useful tool, especially in finding rotor damage or fouling. The borescope can't however, see the splitline or the diaphragm pockets. Borescope inspection of a turbine is equivalent to counting the people in the Superdome while looking through a keyhole. Norco will continue to perform borescope inspections of these turbines between major overhauls.

### *Vibration Monitoring*

These turbines have state of the art online vibration monitoring systems that the people who operate the machines rely on to tell them the condition of the turbine. Machinery engineers worldwide have done a great job selling the operators of the equipment on the value of vibration monitoring. This sometimes works against justifying an overhaul as the operator (who generally controls the budget) feels that as long as the vibration is acceptable, the turbine is in good shape.

Unfortunately, vibration monitoring can generally only reveal the condition of the parts that rotate. As demonstrated in this case, a lot more can go wrong with a turbine than just the parts that rotate. As with borescoping, vibration monitoring is just one tool available in machinery monitoring. In the case of a very insensitive rotor, blade damage and a lost tenion had no noticeable effect on rotor vibration. Complete blade loss is one of the few things that would cause a vibration increase.

### *Efficiency Measurement and Power Loss*

Normal process instrumentation is not accurate enough to measure turbine efficiency. In addition, maintenance of the torque monitors needed to measure efficiency are not a high priority. This all usually means that an efficiency loss is first noticed when the turbine is unable to produce the required amount of power. Performance degradation, while obviously present in this case, was masked by the oversizing of the turbines by about 20 percent. On a more marginal application, this level of damage would have been noticed in the turbines' performance.

The damage found was not detectable by traditional predictive maintenance means such as borescopes, vibration monitoring, or lack of power. The only way to find this sort of damage is to periodically open and inspect the turbine. Obviously, the 13 year run on these turbines was too long. The damage that was found was extensive and would have been much easier to repair had this inspection been performed at an earlier outage. The optimum time interval for these outages has yet to be fully determined since only one data point is available. That data point says that 13 years is too long. It remains to be seen what interval is too short. As a first step, the overhaul interval for these turbines has been changed to every 10 years.

In conclusion, it is the authors' opinion that internal damage will not always manifest itself in a form that is detectable through means other than a full inspection. In today's cost conscious environment, the temptation will always be there to lengthen the inspection interval. As this case shows, there are economic perils in that decision.